ARR Nov. 1941

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

WARTIME REPORT

ORIGINALLY ISSUED
November 1941 as
Advance Restricted Report

PRELIMINARY DATA ON BUCKLING STRENGTH OF

CURVED SHEET PANELS IN COMPRESSION

By Eugene E. Lundquist

Langley Memorial Aeronautical Laboratory
Langley Field, Va.

NACA

WASHINGTON

L-690

PRELIMINARY DATA ON BUCKLING STRENGTH OF CURVED SHEET PANELS IN COMPRESSION

By Eugene E. Lundquist

SUMMARY

This paper presents the results obtained in compression tests of eight stiffened panels. The radius-thickness ratio of the skin between stiffeners varied from 400 to infinity.

From these few tests, it is concluded that the critical compressive stress for a curved sheet between stiffeners is equal to the larger of the following:

(a) The critical compressive stress for an unstiffened circular cylinder of the same radius-thickness ratio

(b) The critical compressive stress for the same sheet when flat

INTRODUCTION

In the design of airplane structures, it is desirable to know the effect of curvature on the strength of the thin metal skin. A number of years ago, the NACA undertook the investigation of the strength of thin-walled duralumin cylinders. The cylinders were tested in a variety of loading conditions, and in all cases failure occurred through instability of the cylinder walls. One of the conditions of loading was that of axial compression. The results of these compression tests are given in reference 1.

As an extension of the cylinder investigation, it was decided to study the critical buckling load for thin curved sheets between stiffeners. The purpose of this report is to present the results of the tests on the first group of specimens in this new investigation.

Ivar C. Peterson, formerly of the NACA staff, performed the experimental work in this investigation.

MATERIALS

The material used in this investigation was 24S-T aluminum alloy. As all specimens were loaded within the elastic range of the material, the modulus of elasticity, E, is the only material property of concern. This value was assumed to be 10,600,000 pounds per square inch in all the calculations of this paper.

SPECIMENS AND THEIR DESIGNATION

The specimens tested in this investigation were made as shown in figure 1. The width of the outstanding flange of each stiffener at the side edges of the sheet, b_F, is given in table I. The different values of b_F were selected so as to force buckling to occur in the sheet between stiffeners and still provide as much support as possible against deflection normal to the sheet.

The angle type of stiffener was selected because of the low rotational restraint that it provided at the side edges of the sheet. The use of two stiffeners at each side edge of the sheet was decided upon in order to stabilize thoroughly these edges against displacements normal to the sheet.

All specimens are designated by their radius-thickness ratio, r/t. In some cases more than one test was made on a given specimen. Thus, the designation of a test is the radius-thickness ratio of the specimen, followed by the number of the test on that specimen. For example, test No. 1318-1 represents the first test on the specimen with r/t = 1318.

In all calculations of stress in this report the areas used were determined from the weight of the specimens. Hence all the dimensions given on figure 1 should be regarded as nominal.

APPARATUS AND METHOD

All specimens were tested in a 300,000-pound compression testing machine. Strains were measured by Tuckerman

optical strain gages of 2-inch gage length on the front and back side of the sheet at each of the locations shown in figure 1. The loading heads of the machine were asjusted so as to bear uniformly on the specimen as indicated by strain gages. As the ends of the specimen had been carefully machined flat, this method of testing was practical although it did take some time to adjust the loading heads of the machine.

No reliable deflection readings were taken in the tests. It was reasoned that the difference in the strain readings on front and back of the sheet was an accurate measure of the change in curvature. According to the theory of small deflections, which applies for all stresses up to the buckling load, the deflection for a given buckled shape is proportional to the curvature. It was considered that the strain readings of the Tuckerman optical strain gages were more accurate than the deflection readings that could be obtained with the equipment at hand.

DETERMINATION OF CRITICAL BUCKLING LOAD

Of the eight specimens tested, those with high curvatures buckled suddenly by a snap diaphragm action, accompanied by a loud report. Those having $r/t = \infty$ and 1318 did not give such a clearly defined critical buckling point. For these latter specimens, there was a gradual growth of deflection that made visual detection of the critical load impossible. In order to obtain the critical buckling load for these two specimens, resort was made to the methods of analyzing experimental observations in problems of elastic stability as given in reference 2. This method is an analysis of the growth of deflection with load and consists of plotting $(y-y_1)/(P-P_1)$ as ordinate against $(y-y_1)$ as abscissa

where

P and y load and corresponding deflection, respectively

P_1 and y_1 initial values of P and y, respectively

The inverse slope of the straight line obtained is $(P_{cr} - P_1)$.

The results of the foregoing type of analysis as applied to the specimen with $r/t = \infty$ and 1318 are given in tables II to V, inclusive, and in figure 2. In this application the symbol y is the difference in strain readings between the two strain gages at the center of the sheet.

RESULTS

The results of these tests are given in table VI and figures 3 and 4. Figure 3 has been prepared to show how the strains at the middle of the sheet varied below the critical buckling loads. It is observed from figure 3 that in each test the two strain gages on the front and back of the sheet reveal a gradually increasing difference of strain as the critical load is approached. For the specimens with $r/t = \infty$ and 1318, the gradual increase in the difference of strain continued through the critical load, whereas for all other specimens there was an abrupt change in the strain readings at the critical load caused by the snap diaphragm action. As the intensity or force of the snap diaphragm action increased with increasing curvature, the strain gages were removed just before the buckling load was reached in the case of the specimens with $r/t = 478$, 432, and 400.

In order to study the effect of the radius-thickness ratio, r/t, on the critical strain, f_{cr}/E, figure 4 was prepared. The lines labeled A, B, and C on this figure are the same as lines A, B, and C in figures 7 and 9 of reference 1. The horizontal lines labeled $r/t = \infty$ in figure 4 give the experimentally determined critical strain for the flat specimen of this investigation. The points plotted on figure 4 give the observed critical strain on the remainder of the specimens with each point plotted at the r/t value for the particular specimen represented by that point.

From the test data plotted in figures 7 and 9 of reference 1, it is clear that line B in figure 4 of this paper gives the upper limit of the experimentally determined values of f_{cr}/E for thin-walled cylinders without longitudinal stiffeners. The fact that the test points in figure 4 plot along line B and the horizontal lines labeled $r/t = \infty$, indicates that the critical compressive stress for a curved sheet between stiffeners is equal to the larger of the following:

(a) The critical compressive stress for an unstiffened circular cylinder of the same radius-thickness ratio

(b) The critical compressive stress for the same sheet when flat

When these tests were planned it was expected to find that in all cases curvature would raise the critical compressive stress over that for the flat specimen $r/t = \infty$. (See equation (276), p. 470, of reference 3.) Why the experimental point for the specimen with $r/t = 806$ fails to check standard theory in this respect (see fig. 4) has not yet been explained. Additional tests are being made in the course of a further study of this point.

CONCLUSIONS

From these few tests, it is concluded that the critical compressive stress for a curved sheet between stiffeners is equal to the larger of the following:

(a) The critical compressive stress for an unstiffened circular cylinder of the same radius-thickness ratio

(b) The critical compressive stress for the same sheet when flat

Langley Memorial Aeronautical Laboratory,
National Advisory Committee for Aeronautics,
Langley Field, Va.

REFERENCES

1. Lundquist, Eugene E.: Strength Tests of Thin-Walled Duralumin Cylinders in Compression. NACA Rep. No. 473, 1933.

2. Lundquist, Eugene E.: Generalized Analysis of Experimental Observations in Problems of Elastic Stability. NACA TN No. 658, 1938.

3. Timoshenko, S.: Theory of Elastic Stability. McGraw-Hill Book Co., Inc., 1936.

L-690

TABLE II.- TEST ∞ -1

P (lb)	y (in.)	$P-P_1$ (lb)	$y-y_1$ (in.)	$\frac{y-y_1}{P-P_1}$ (in./lb)
200	0	0	0	-
1000	.000002	800	.000002	0.2500×10^{-8}
2000	.000004	1800	.000004	.2222
3000	.000008	2800	.000008	.2857
4000	.000016	3800	.000016	.4211
5000	.000032	4800	.000032	.6667
6000	.000060	5800	.000060	1.0345
6500	.000094	6300	.000094	1.4921
7000	.000142	6800	.000142	2.0882

TABLE III.- TEST ∞ -2

P (lb)	y (in.)	$P-P_1$ (lb)	$y-y_1$ (in.)	$\frac{y-y_1}{P-P_1}$ (in./lb)
200	0	0	0	-
1000	.000002	800	.000002	0.2500×10^{-8}
2000	.000004	1800	.000004	.2222
3000	.000004	2800	.000004	.1429
4000	.000012	3800	.000012	.3158
5000	.000022	4800	.000022	.4583
5500	.000030	5300	.000030	.5660
6000	.000046	5800	.000046	.7931
6500	.000072	6300	.000072	1.1429
7000	.000112	6800	.000112	1.6471
7250	.000132	7050	.000132	1.8723

TABLE IV.- TEST ∞ -3

P (lb)	y (in.)	$P-P_1$ (lb)	$y-y_1$ (in.)	$\frac{y-y_1}{P-P_1}$ (in./lb)
500	0	0	0	-
1000	.000020	500	.000020	4.000×10^{-8}
2000	.000018	1500	.000018	1.200
3000	.000020	2500	.000020	.800
4000	.000026	3500	.000026	.743
4500	.000032	4000	.000032	.800
5000	.000036	4500	.000036	.800
5500	.000048	5000	.000048	.960
6000	.000070	5500	.000070	1.273
6500	.000108	6000	.000108	1.800
6750	.000126	6250	.000126	2.016
7000	.000154	6500	.000154	2.369

TABLE I.- DIMENSIONS OF SPECIMENS

$\frac{r}{t}$	Radius of sheet (in.)	Outstanding flange of angle stiffener, b_F (in.)
∞	∞	1.51
1318	103.0	1.51
806	63.3	1.08
634	49.5	1.08
542	42.2	.87
478	37.3	.87
432	33.8	.75
400	31.1	.75

TABLE V.- TEST 1318-1

P (lb)	y (in.)	$P-P_1$ (lb)	$y-y_1$ (in.)	$\frac{y-y_1}{P-P_1}$ (in./lb)
500	0	0	0	-
1000	.000002	500	.000002	0.4000×10^{-8}
2000	-.000002	1500	-.000002	-.1333
3000	.000012	2500	.000012	.4800
4000	.000012	3500	.000012	.3429
4500	.000018	4000	.000018	.4500
5000	.000020	4500	.000020	.4444
5500	.000026	5000	.000026	.5200
6000	.000040	5500	.000040	.7273
6500	.000068	6000	.000068	1.1333
6750	.000086	6250	.000086	1.3760
7000	.000110	6500	.000110	1.6923
7250	.000142	6750	.000142	2.1037

TABLE VI.- TEST RESULTS
$(E = 10.6 \times 10^6)$

Test	$\frac{r}{t}$	Area, A (sq in.)	Buckling load, P_{cr} (lb)	Buckling stress, f_{cr} (lb/sq in.)	$\frac{f_{cr}}{E}$
∞ -1	∞	1.3100	7,430	5,670	5.349×10^{-4}
∞ -2	∞	1.3100	7,630	5,820	5.491
∞ -3	∞	1.3100	7,660	5,850	5.519
1318-1	1318	1.3100	7,670	5,850	5.519
806-1	806	1.1578	6,250	5,400	5.094
634-1	634	1.1552	7,800	6,750	6.368
634-2	634	1.1552	7,910	6,850	6.462
542-1	542	1.1029	7,280	6,600	6.226
478-1	478	1.1020	9,030	8,190	7.726
432-1	432	1.0668	9,380	8,790	8.292
400-1	400	1.0668	11,050	10,360	9.774

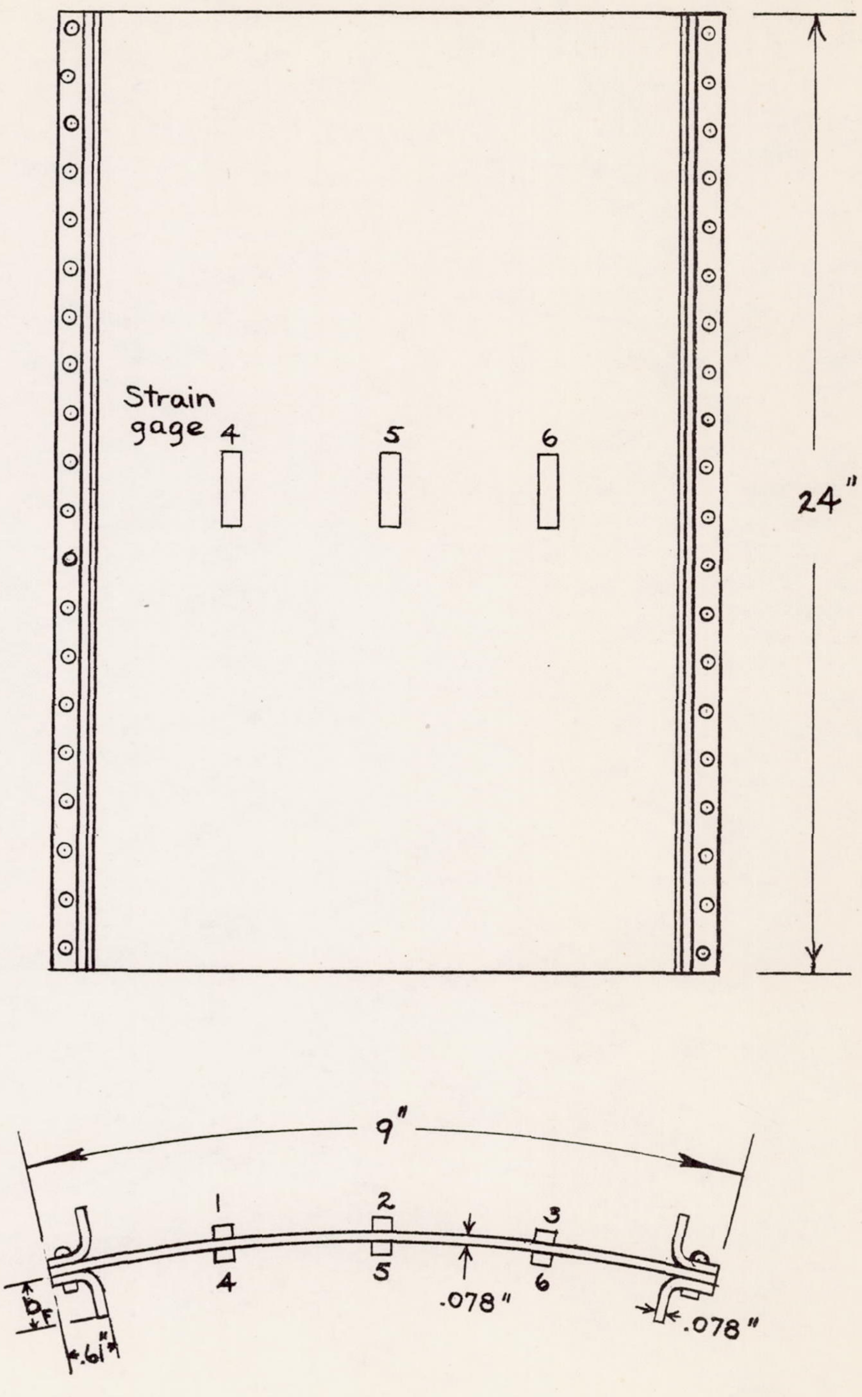

Figure 1. - Test specimen.

L-690

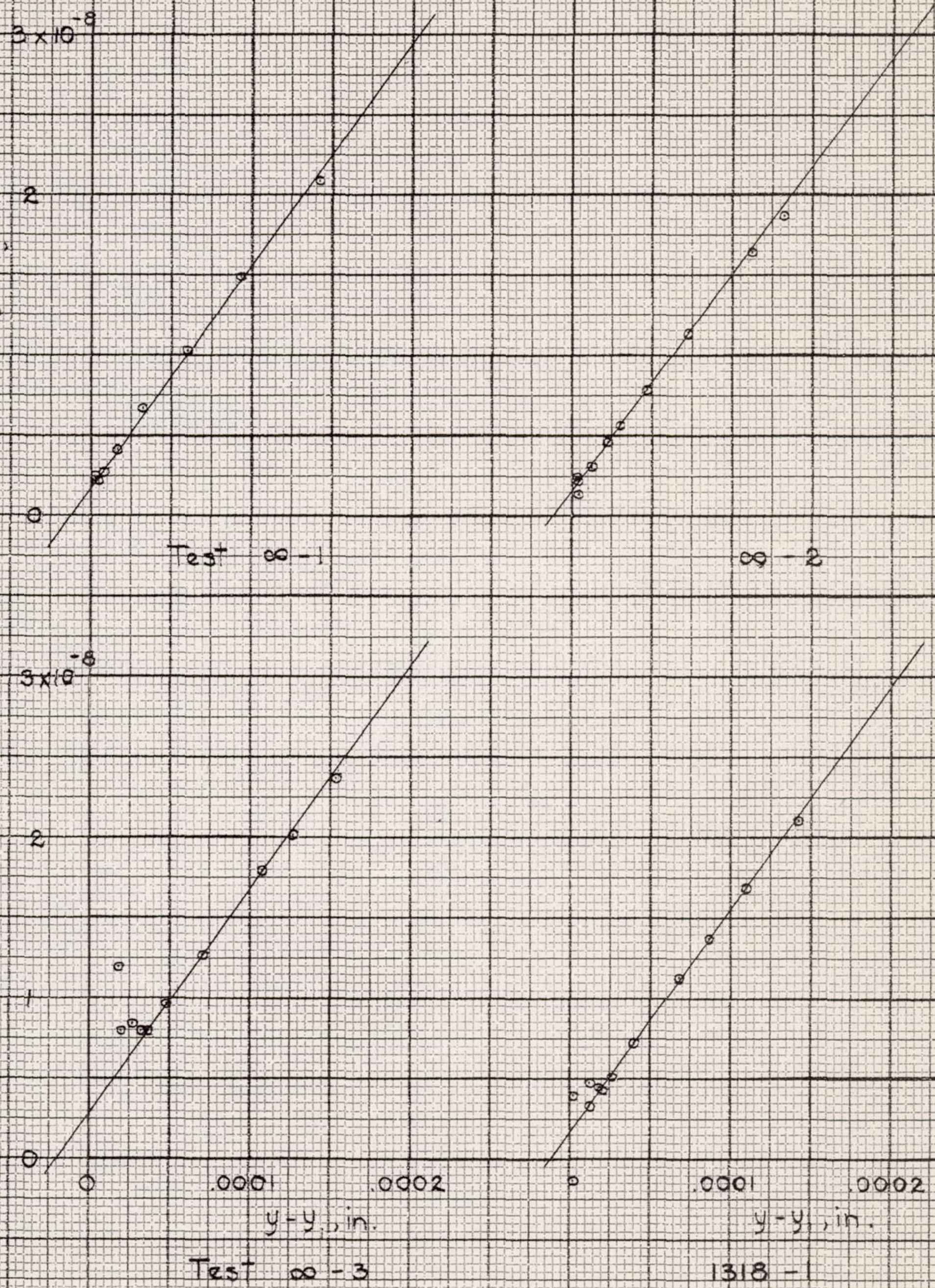

Figure 2.- Load-curvature data for prediction of critical loads.

L-690

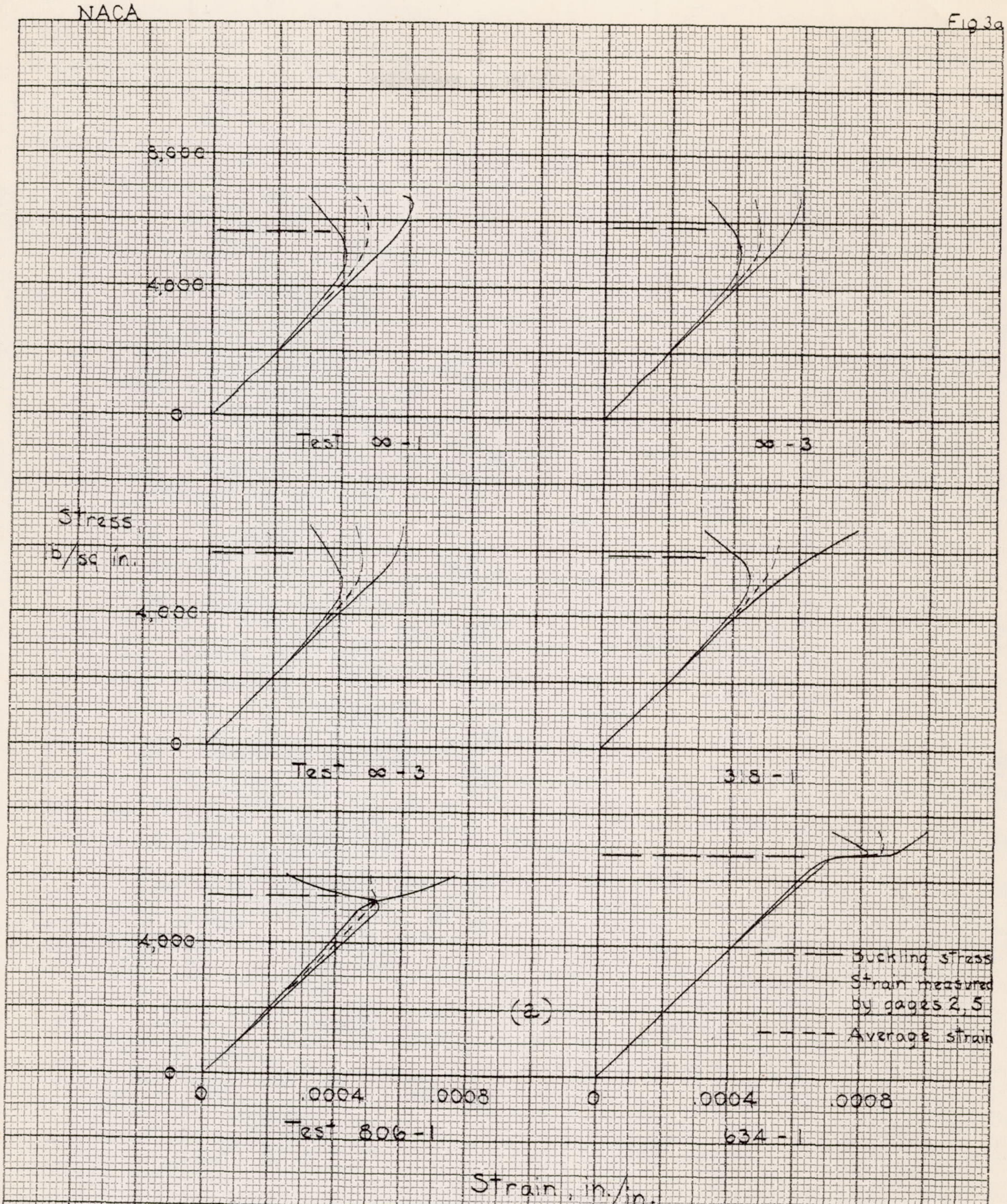

Figure 3ab-Average stress on specimen against strain readings taken at the middle of the specimen.

L-690

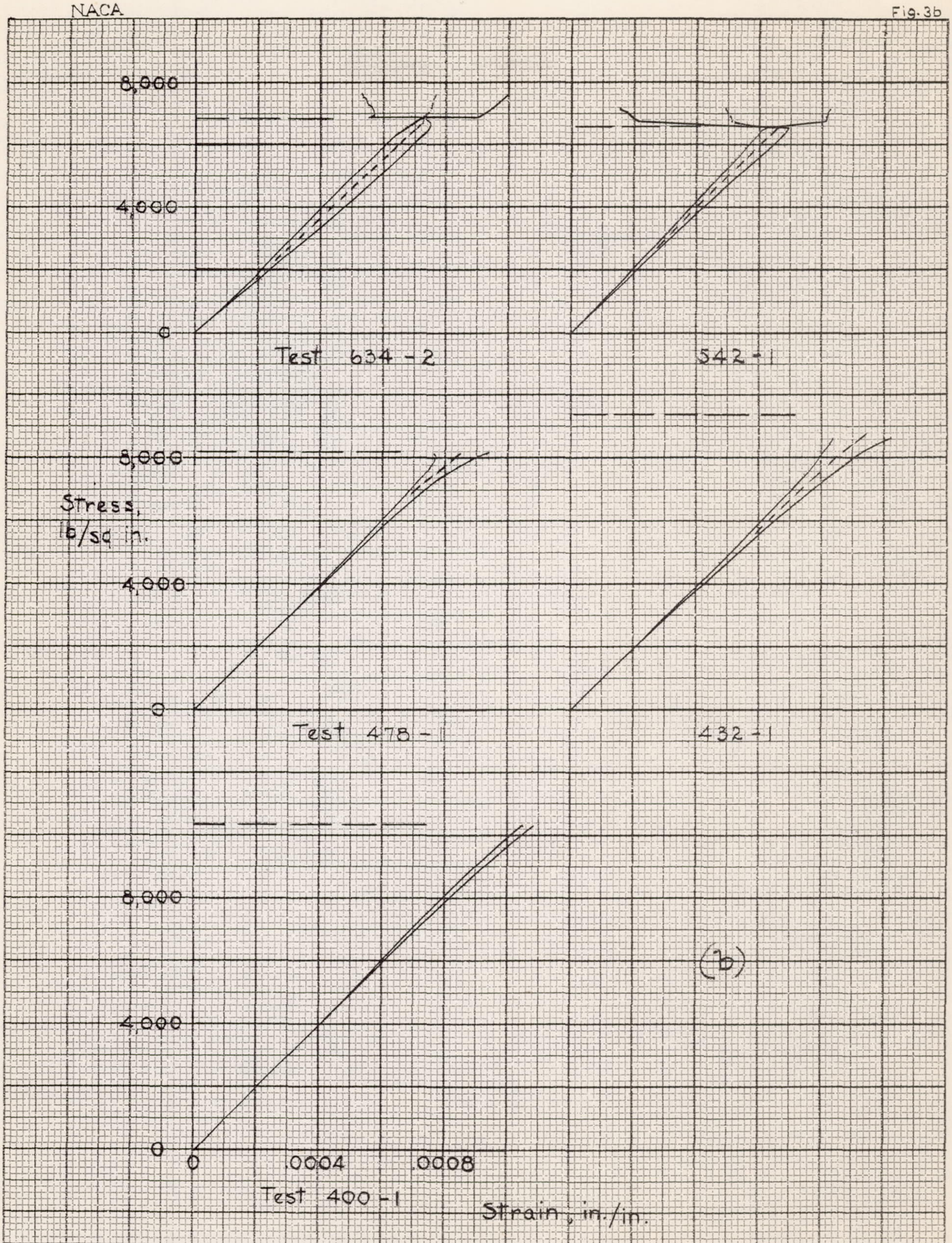

Figure 3. - concluded

L-690

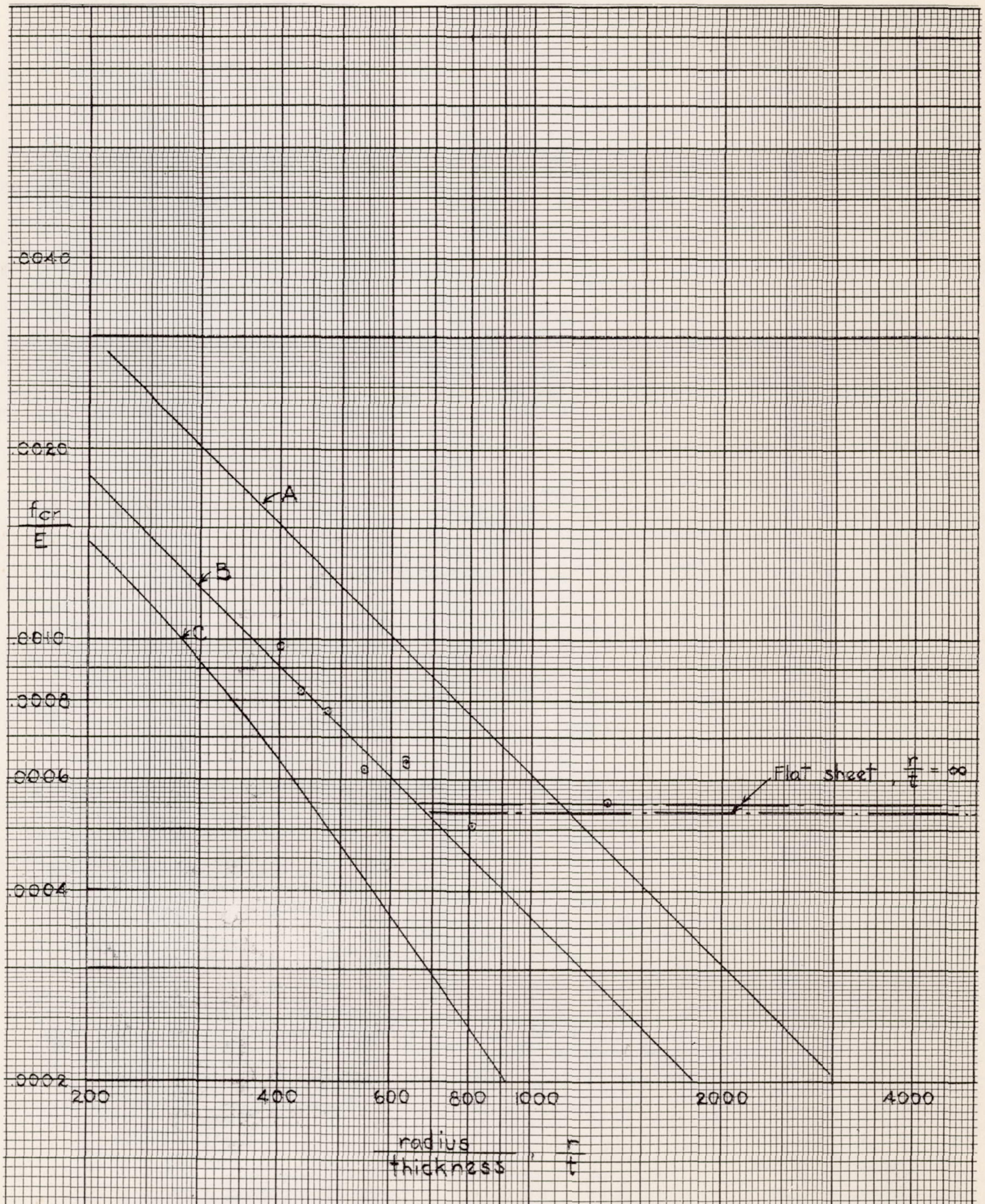

Figure 4.- Critical compressive stress for 24 S-T aluminum alloy curved sheet between stiffeners.